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To cite this article: Y Zhang *et al* 2021 *IOP Conf. Ser.: Mater. Sci. Eng.* **1180** 012060

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Simulation and optimisation of opened sewage concentration air cycle combined with the heat pump under actual weather conditions

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Abstract. Opened air cycle is a promising sewage concentration technology using heated air as the medium to absorb water from the sewage. Unlike closed cycles, the air flow is directly ejected out of the system after humidification in opened cycles. Operating conditions and the cycle's efficiency are highly affected by the weather, such as the time-varying ambient temperature and relative humidity. Since the evaporating temperature relies on the ambient temperature, to restrain its dependence, the evaporator was proposed combined with the ejected air flow in this paper. Such a design could improve the efficiency of the heat pump by increasing its evaporating temperature. Simulation of the newly proposed open sewage concentration cycle was performed to investigate its thermal process and economic efficiency under actual weather conditions. Thermodynamic models of the compressor, heat exchangers and the sewage concentration chamber were established. Climatic conditions at Xi'an, China were referenced as actual weather conditions. Optimisation of the working parameters was performed to achieve higher economic index. The results demonstrated that the average monthly power consumption of the optimized cycle was 0.3508 kWh/kg. Compared with the published data of the closed cycle, the total cost of new opened cycle was nearly one-third lower.

1. Introduction

With the development of industry, sewage treatment demands are increasing year by year. It will cause environmental pollution and water waste if the sewage is charged directly. The study on economic and efficient wastewater concentration technology is important to China's industrial circles^[1,2]. As a kind of efficient and energy-saving wastewater concentration technology, heat pump drying technology solves the problem of difficult structural design and easily fouling^[3] in MVR^[4,5] and membrane process^[6]. It can be applied in atmospheric pressure and low ambient temperature, which has obvious advantages for heat sensitive sewage^[7,8]. So such technology has broad prospects for application in actual industry.

In 1943, Sulzer Company first used heat pump drying technology to build dehumidification drying device^[9]. Xie proposed closed liquid concentration cycle combined with the heat pump under atmospheric pressure, which provided the possibility of efficient and economic concentration of heat-sensitive material liquid^[10]. Wang applied this technology to the concentration of copper pipe washing



solution, which could recover 70% water in waste liquid with low energy consumption, proving the practicability of this technology^[11]. Liu added bypass pipe device to the closed liquid concentration system to save energy, which successfully reduced energy consumption by 30%^[12]. Saensabai conducted system simulation calculations theoretically on both opened and closed structures, and found that the theoretical energy consumption ratio of the opened structure was lower^[13]. This was because in the closed cycle, circulating air needed to be dehumidified in the evaporator in order to assure the good moisture absorption later in the sewage concentration chamber. So the evaporating temperature would be lower than that of opened structures under similar operation conditions, causing the system more power-wasting. So in order to build an opened structure, the evaporator was proposed combined with the ejected air flow in this paper.

In the opened system, Yuan analyzed the influence of evaporation pressure according to the ideal model^[14]. Zhou conducted an experimental investigation of the effects of spray flow rate, circulating air flow rate and trade effluent concentration on the treatment capacity^[15]. Zhang studied the relationship between the gas-liquid mass flow ratio, waste liquid temperature and the concentration efficiency of the device in order to estimate the performance of the system^[16]. Later, Zhang proposed the experimental correlation formula of wastewater concentration efficiency, and the maximum error between the calculated value and the experimental value was within 20%^[17]. The research on opened liquid concentration system mainly focused on experiments and ideal model analysis. Realistic simulation and seasonal optimisation of systems under actual weather conditions still need further study.

The main aim of this study is to assess the thermal process and economic efficiency of an opened sewage concentration air cycle combined with the heat pump under actual weather conditions. The improved cost annual value method is applied to evaluate the investment cost, maintenance cost and operating cost of the system comprehensively. The monthly optimisation of the actual system mainly depends on operation parameters such as time-varying ambient temperature and relative humidity, the evaporating temperature and the condensing temperature of the heat pump. The influences of these operation parameters on key parameters like inlet air flow, evaporation mass flow and power consumption are studied.

2. Mathematical model of the system

2.1. System description

Opened sewage concentration air circle includes heat pump loop, air loop and sewage loop. The main process of the circle designed in this paper is shown in figure 1.

The ambient air flows through the condenser after preheating. With higher temperature and lower relative humidity, the air flow is pumped into the sewage concentration chamber by the fan. In the chamber, the wet air is hygroscopic and cooled, then becomes saturated wet air^[15]. The heat is recovered by the heat exchanger 2. After separating condensate water, the air flow is directly ejected by the evaporator.

The heat exchange temperature difference ΔT is settled as 5°C. The system pressure P_{sys} is 101.325 kPa. According to Li's thesis^[18], the wind speed is set as 15 m/s in which case the evaporation efficiency is larger and there will not be obvious liquid floating phenomenon. The mass ratio of inlet sewage to concentrated sewage is 6:1. emoney (the electricity consumption of evaporated water per unit mass) is introduced as the measurement standard of thermal efficiency of the system. emoney is equal to the total electricity consumption of all energy-consuming components (including compressor, fan, pump, etc.) divided by m_{zfph} (total evaporation mass rate of the sewage in the system).

2.1.1. Heat pump loop and fan. The heat pump loop is the heat energy source of the whole system. It uses R134A as the working medium. By optimizing the evaporating temperature and condensing temperature of the heat pump, the compressor power can be calculated.

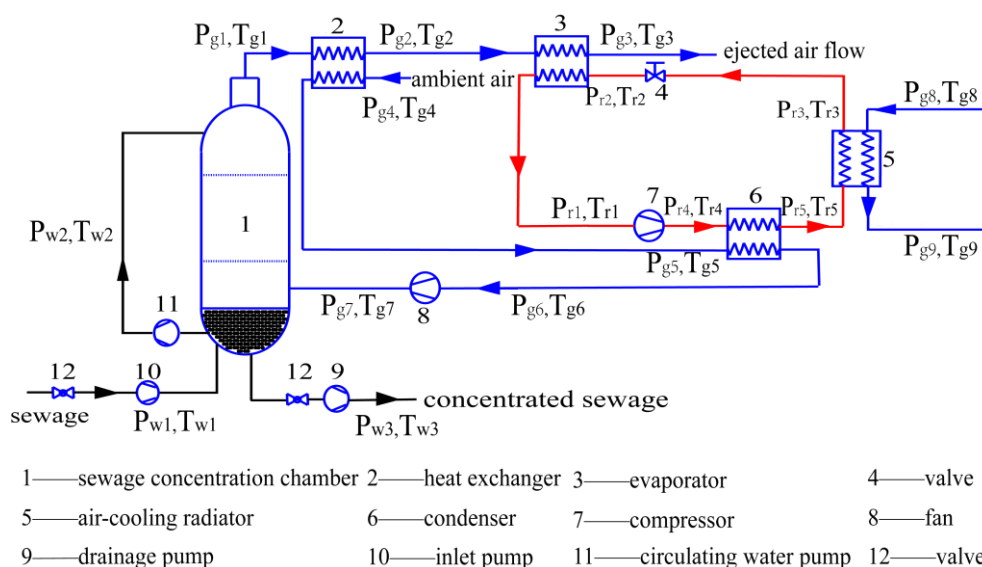


Figure 1. Flow chart of opened sewage concentration air cycle combined with the heat pump.

The fan drives the air flow of the system. The total fan power needs to consider the resistance loss of the gas-liquid separator, heat exchangers and pipeline flow resistance in the circle. The resistance losses of gas-liquid separators are mostly determined by experiments, and lacks the correlation formula. Therefore, the resistance loss of gas-liquid separators is set as general number 10 kPa^[19]. The loss calculation of heat exchanger, pipeline resistance^[20,21] and airflow resistance in the chamber are shown in table 1.

In summary, N_{fan} (the total fan power) can be calculated as equation (1).

$$N_{fan} = Q_{air} (P_{local} + P_{onway} + P_{ht} + P_{fan} + P_{sys}) / \eta_{fan} \quad (1)$$

Where Q_{air} is the air flow rate in the pipe; the meaning of P_{local} , P_{onway} , P_{ht} and P_{fan} is shown in the table 1; P_{sys} is system pressure. η_{fan} is the fan efficiency.

Table 1. The loss calculation of pipes, heat exchangers and the chamber.

Physical meaning	Calculation formula
Pipeline resistance loss along the way P_{onway}	$P_{\text{onway}} = \frac{\delta}{d} \times \frac{v^2 \rho}{2} l$
Local resistance loss of pipeline P_{local}	$P_{\text{local}} = 0.5 \xi \rho v^2$
Total friction resistance of heat exchanger P_{htf}	$P_{\text{htf}} = P_{\text{htfl}} \times \Phi_1^2$
Liquid phase shunt resistance of heat exchanger P_{htfl}	$P_{\text{htfl}} = 2 f_1 L G^2 (1 - x^2) v_1 D^{-1}$
Coefficient of liquid friction along the heat exchanger f_1	$f_1 = 2.5 Re^{-0.3} \quad [22]$
Gas phase shunt resistance of heat exchanger P_{htfg}	$P_{\text{htfg}} = 2 f_g L G^2 x^2 v_g D^{-1}$
Coefficient of gas friction along the heat exchanger f_g	$f_g = 0.3164 Re^{-0.25} \quad [23]$
Friction component liquid phase apparent coefficient Φ_1^2	$\Phi_1^2 = 1 + CX^{-1} + X^{-2}$
Martinelli parameter X	$X^2 = P_{\text{htfl}} \times P_{\text{htfg}}^{-1}$
Total resistance loss of heat exchanger P_{ht}	$P_{\text{ht}} = (1.1 - 1.15) P_{\text{htf}}$
Air resistance loss in the the chamber P_{fan}	$P_{\text{fan}} = 0.5 \varepsilon v^2 \rho \quad [24]$

2.1.2. Sewage concentration system. The flow chart of the evaporation system is also shown in the left part of figure 1. The main components include the sewage concentration chamber and pumps.

In this paper, air is directly heated and contacted with cold sewage in the chamber, which can avoid fouling problem in the heat exchanger. For the sewage concentration chamber, the chamber temperature T_{g1} is calculated through equations of mass and energy conservation.

The inlet sewage temperature is set as normal temperature 20°C in summer and 15°C in winter. The evaporation mass rate m_{zfph} is calculated by equation (2).

$$m_{zfph} = m_{air} (d_{g1} - d_{g6}) \quad (2)$$

Where m_{zfph} is the evaporation mass rate of the system; m_{air} is the mass flow rate of dry air; d_{g1} and d_{g6} are the moisture content of point $g1$ and $g6$ respectively.

When calculating circulating pump power, the pressure loss of fluid flowing through the nozzle is considered. The calculation formulae are shown in table 2.

Table 2. Calculations of circulating pump power.

Physical meaning	Calculation formula
The pump power q_{pump}	$q_{pump} = 9.8HQ(n_1n_2)^{-1}$ [25]
Injection drop ΔP	$\Delta P = 0.5\rho Q^2 C^{-2} A^{-2}$ [26]
Total pump power q_{pumpt}	$q_{pumpt} = \Delta P \times Q + q_{pump}$

When the circle runs steadily, the air has reached saturation state out of the sewage concentration chamber and the saturation temperature under this pressure is not higher than the temperature of the sewage concentration chamber. In addition, heat and mass transfer driven by the pressure of water vapor in the chamber is weakened after reaching the saturation state. The dominant driving power turns into the temperature difference between sewage and air. Therefore, the temperature of air in the chamber needs to be higher than that of sewage to ensure the stable operation of the system.

2.2. Cost calculation

In this paper, the improved cost annual value method is applied for analysis. The cost of evaporated water per kilogram COST is calculated by equation (3).

$$COST = EC + \frac{IC \times CRF}{WT \times m_{zfph}} + \frac{MC}{WT \times m_{zfph}} \quad (3)$$

Where selected parameter values in the model are shown in table 3.

2.2.1. Cost calculation of Electricity energy. The electricity energy includes the energy consumed by components like the compressor, the fan, and the pump. The electricity bill EC is calculated as equation (4).

$$EC = ele_{price} \times emoney \quad (4)$$

Where ele_{price} is the electricity price.

Table 3. Parameter values in the economic model of the actual circle.

Parameter	Values
System investment return years/yr	5
The discount rate /%	10
Investment recovery coefficient CRF	0.264
Electricity price $ele_{price}/\text{yuan} \cdot \text{kWh}^{-1}$	0.8675
Annual running time of the system WT/h	8600

2.2.2. Cost calculation of initial investment. The initial investment IC includes equipment costs, installation cost, labour cost and construction cost.

According to ‘Estimated Index of National Municipal Engineering Investment’^[27], the cost is calculated as equation (5).

$$IC = 1.178 \sum C_i \quad (5)$$

Where C_i is the price of the key equipment.

Table 4 shows the price calculation of the key equipment in the system.

Table 4. Price calculation of the key equipment.

Equipment	Purchase cost calculation
Compressor	$1.051 \times 39.5 m_{\text{ref}} \times \varepsilon \ln \varepsilon \times (0.9 - \eta_{\text{ad}})^{-1}$ [28]
Heat exchanger	$2143 \times A_{\text{ht}}^{0.514}$ [29]
Throttle valve	$114.5 m_{\text{air}}$ [30]

2.2.3 Cost calculation of maintenance. According to RAZMI’s paper, the maintenance cost is calculated as equation (6) [31].

$$MC = 1\% IC \quad (6)$$

Where the maintenance fee MC includes the management fee and staff salaries.

3. System simulation

3.1. Weather data

The opened sewage concentration air cycle to be evaluated in this study is assumed to be installed in Xi 'an. Figure 2 shows the average air temperature and relative humidity data of Xi 'an.

In the simulation, *emoney* (power consumption of evaporating water per unit mass) was selected as the optimisation target of the system. *emoney* may be affected by ambient temperature and relative humidity, as well as the evaporating and condensing temperatures of the heat pump. Since the ambient temperature and relative humidity are given, evaporating temperature and condensing temperature are chosen as optimisation variables to reach the monthly minimum of *emoney*. In this way, a four-dimensional problem is successfully simplified to a two-dimensional problem.

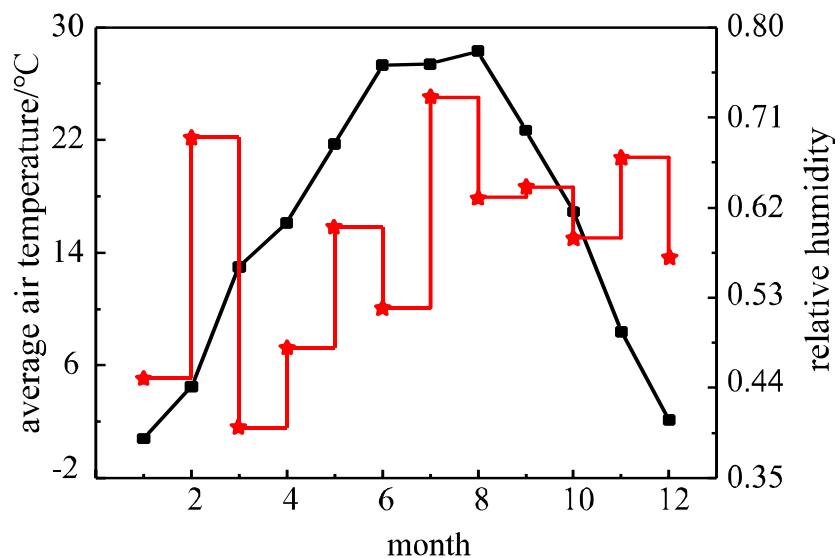


Figure 2. Air temperature and relative humidity in Xi 'an.

3.2. Optimisation model of the actual operating system

The refrigerating capacity is determined by the working conditions of the compressor. When calculating the compressor power, the adiabatic efficiency and volume efficiency are involved.

The evaporation mass of the system is no longer fixed. The air flow rate is adjusted to balance the heat exchange between the air side and the refrigerant side in heat exchangers.

The model can be expressed by equation (7). The complex method is used to solve the problem and get the corresponding monthly optimal operating conditions.

$$\min emoney(T_{evp}, T_{con})$$

$$s.t. \begin{cases} 21 \leq T_{con} < 1.607T_{evp} + 74.1, -15 \leq T_{evp} < 0 \\ 21 \leq T_{con} < 74.1, 0 \leq T_{evp} < 7 \\ 1.25T_{evp} + 11.25 \leq T_{con} \leq 74.1, 7 \leq T_{evp} \leq 15 \end{cases} \quad (7)$$

Where T_{con} is the condensing temperature and T_{evp} is the evaporating temperature.

4. Results and discussion

4.1. Influencing factors of system performance

By calculation, the change of air relative humidity has small influence on the system performance compared with other variables, so this paper mainly studies on the influence of other three variables: inlet air temperature, evaporating temperature and condensing temperature.

4.1.1. Influences of inlet air temperature on system performance. Figure 3 displays change rules of key parameters in the system such as inlet air flow rate and evaporation mass flow rate when inlet air temperature is changed.

When evaporating temperature, condensing temperature is constant, refrigerant states and refrigerant mass flow rate in the heat pump loop stay constant, so the refrigerating capacity and compressor power consumption are not changed. The chamber temperature rises with the air temperature, and inlet air flow rate decreases due to constant refrigerating capacity, which leads to reduced evaporation mass flow, causing *emoney* increasing.

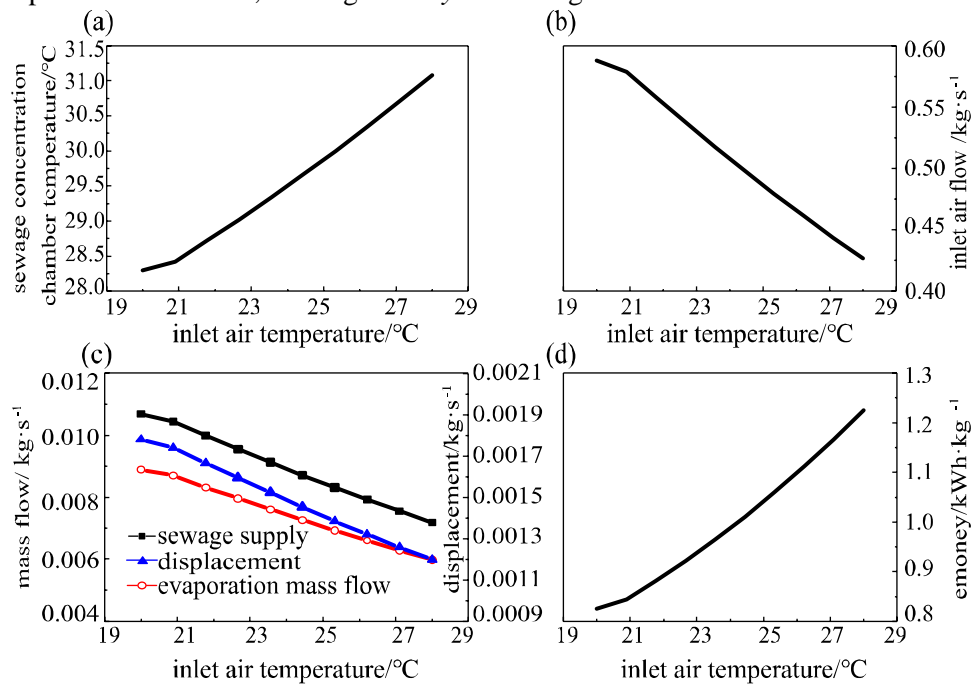


Figure 3. Influences of inlet air temperature on key parameters of the system: (a) sewage concentration chamber temperature; (b) inlet air flow; (c) m_{zfp} , sewage supply and displacement; (d) *emoney*.

4.1.2. Influences of heat pump evaporating temperature on system performance. Figure 4 displays change rules of key parameters in the system such as inlet air flow rate, evaporation mass flow rate and power consumption when the evaporating temperature is changed.

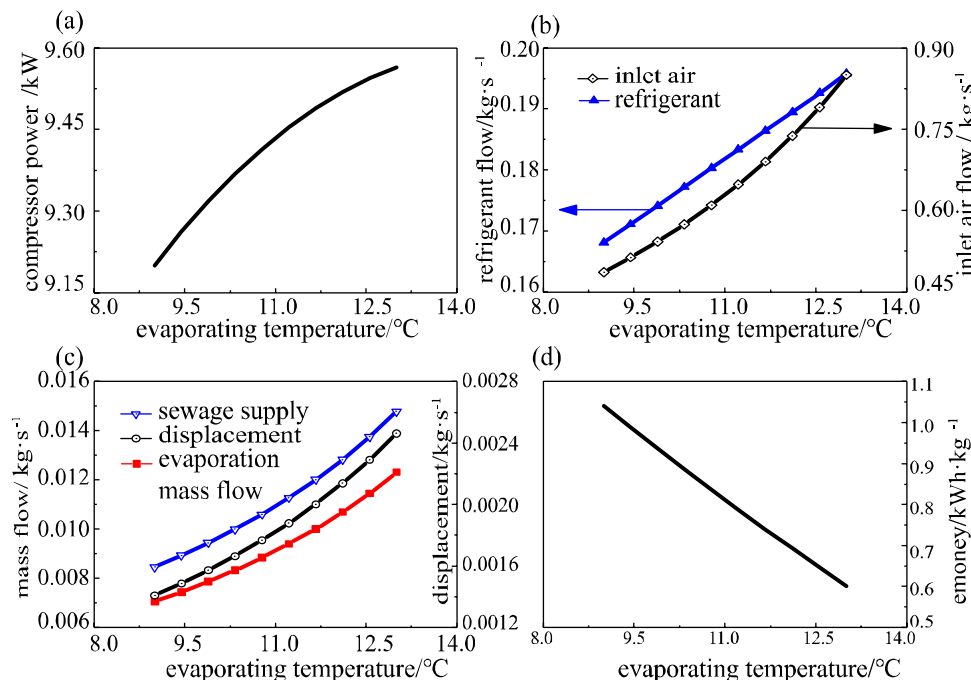


Figure 4. Influences of evaporating temperature on key parameters of the system: (a) compressor power; (b) refrigerant and inlet air flow; (c) m_{zfp} , sewage supply and displacement; (d) $emoney$.

For the refrigerant cycle, when the condensing temperature is constant and the evaporating temperature is increased, the density of inlet gas of the compressor is increased. The volumetric efficiency of the compressor is also increased, which leads to the increased mass flow rate of the refrigerant. So the compressor power consumption rises up. The total power consumption of other dissipative components is around 12 kW and only changes a little. So total power consumption rises with the evaporating temperature. The inlet air flow rate increases, resulting in an increase in sewage supply, evaporation mass flow and displacement. The increasing speed of system evaporation mass is faster than that of power consumption, so $emoney$ decreases.

4.1.3. Influences of heat pump condensing temperature on system performance. Figure 5 displays change rules of key parameters in the system such as evaporation mass flow rate and power consumption when the condensing temperature is changed.

When the condensing temperature increases, the outlet temperature of the condenser increases, leading to the increase of the chamber temperature. For the heat pump loop, the increase of pressure ratio leads to the decrease of compressor volumetric efficiency, while the evaporation temperature remains the same, so the density of the inlet gas through the compressor is invariant. The refrigerant mass flow rate is equal to the inlet gas density times theoretical volume times volume efficiency. So the refrigerant mass flow rate drops, leading to a decline in refrigerating capacity. Therefore, the inlet air mass flow rate drops, resulting in a decrease in water supply, displacement and evaporation mass flow. The compressor power consumption increases, causing total power consumption rising, so $emoney$ increases.

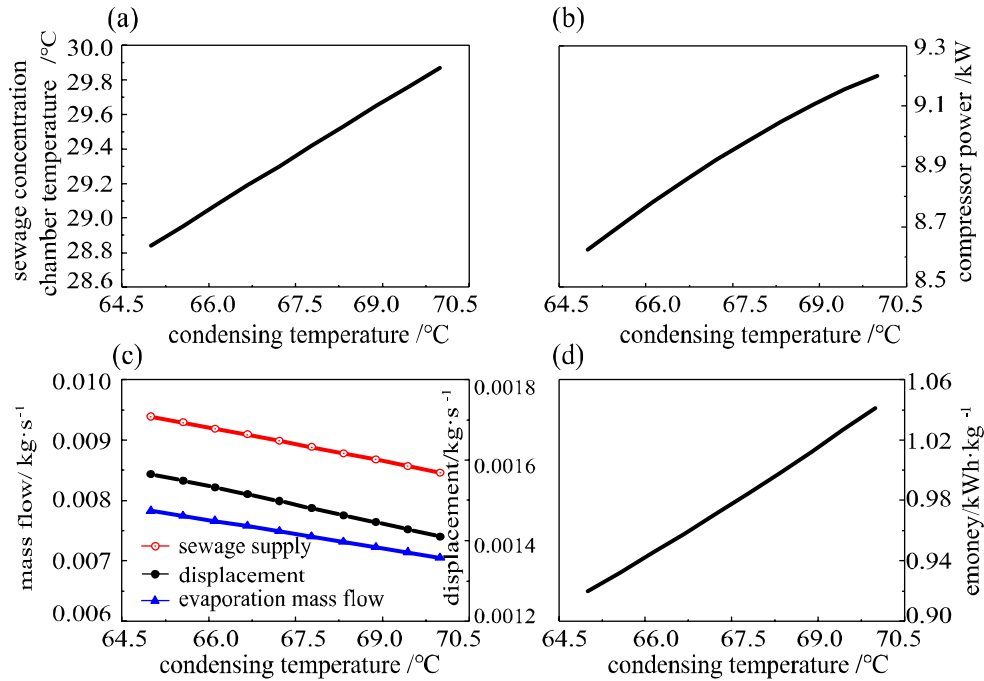


Figure 5. Influences of condensing temperature on key parameters of the system: (a) sewage concentration chamber temperature; (b) compressor power; (c) m_{zfp} , sewage supply and displacement; (d) $emoney$.

4.2. Energy consumption and water treatment

According to the analysis of the influence of the above operating parameters on the energy consumption of the system, the direction of performance optimisation of the system can be concluded. Figure 6 displays calculated $emoney$ and evaporation mass flow of each month in the corresponding optimal operating conditions under actual weather conditions.

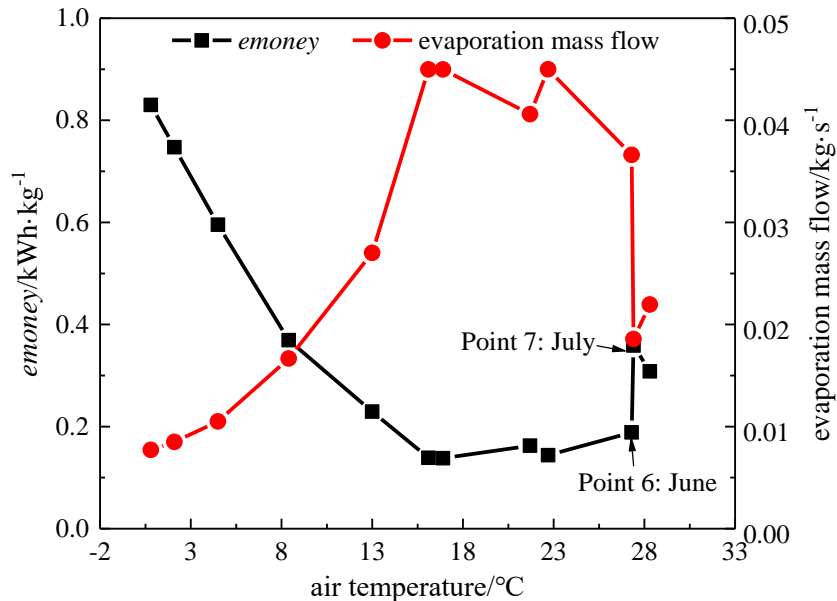


Figure 6. $emoney$ and evaporation mass flow under actual weather conditions.

The average monthly power consumption of evaporated water per unit mass is 0.3508 kWh/kg. At low temperatures, the evaporating temperature becomes quite low due to the limit of the cold air

temperature, which causes a surge in compressor power consumption and a decline in evaporation mass flow. And finally *emoney* is expected to soar. At high temperatures, the overall operating performance is at the high level. The decrease of evaporation mass flow leads to the increase of *emoney*. To avoid the phenomenon of liquid floating, the inlet air flow rate needs to be limited at high temperatures, leading to the decline of evaporation mass flow and the rise of *emoney*. The temperature difference of point 6 and point 7 is 0.1°C while the relative humidity difference is 21%. When temperatures are similar, *emoney* increases with increasing air relative humidity due to the rise in compressor power.

4.3. Economic analysis

Figure 7 shows *COST* under monthly optimized operating conditions can be calculated based on the economic model above.

As seen from Figure 7, *COST* reaches a minimum value in April when its temperature is 16.1°C . The tendency of *COST* changes correspondingly in agreement with *EC* (the electricity cost). *IC* and *MC* changes little when the actual system is settled. *COST* jumps in July because the electricity cost (is proportionate to *emoney*) rises up in July. The compressor power plays a leading role in the electricity cost at low temperatures while the evaporation mass flow is dominant in the change tendency of *emoney* at high temperatures.

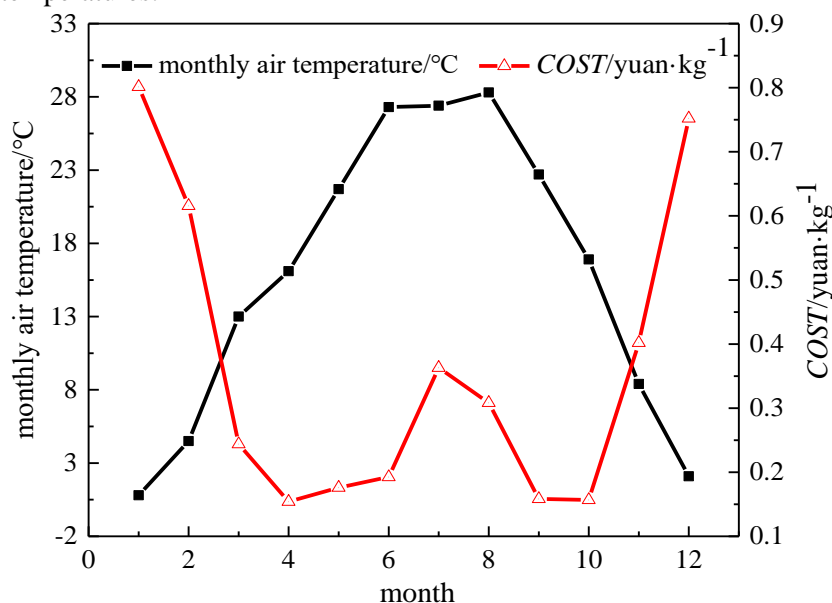


Figure 7. *COST* under monthly optimized operating conditions.

The calculated *COST* (including investment cost, electricity cost and maintenance cost) is 209,300 yuan/yr, evaporation mass flow rate is 833.6 t/yr, sewage treatment is 1000.3 t/yr, concentrated sewage is 166.714 t/yr and average evaporated water per ton costs 251.1 yuan/t.

According to the data provided in Xie's paper^[32], the daily evaporated water mass is 50kg and the operation cost is about 6570 yuan/yr, so the average evaporated water cost is 360 yuan/t. So the total cost of the new opened cycle in this paper is nearly one-third lower.

5. Conclusions

The main purpose of this paper is to investigate the thermal process and economic efficiency of the opened sewage concentration air cycle under actual weather conditions. Based on the economic index, the system is optimized monthly and the following conclusions are drawn:

(1) When the evaporating temperature and condensing temperature are constant, the increase of inlet air temperature may decrease inlet air flow and evaporation mass flow, causing a drop in *emoney*.

(2) When temperatures are similar, *emoney* increases with increasing air relative humidity, causing a rise in *COST*.

(3) For the actual system, the average monthly power consumption of evaporated water per unit mass is 0.3508kwh /kg, which is relatively energy-saving.

(4) By optimizing the system monthly, the average the average evaporated water cost under actual weather conditions is 30.3% lower compared with the cost of the closed system in the literature.

The results of the analyses indicate that opened sewage concentration air cycle combined with heat pump is economically feasible for sewage concentration. This study may be improved in future work with the addition of an energy recovery device.

Acknowledgements

This paper is supported by General Project of Chinese National Natural Fund (Project No.: 52076162).

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